DYNAMICS OF DRIVE SYSTEMS FOR WIND ENERGY CONVERSION

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ABSTRACT

Calculations are performed to determine the dynamic effects of mechanical power transmission from the nacelle of a horizontal axis wind machine to the ground or to an intermediate level. It is found that resonances are likely at 2 or 4/REV, but they occur at low power only, and seem easily correctable. Large reductions are found in the harmonic torque inputs to the generator at powers near rated.

Introduction

Due to the large weight of the gear-up and generation equipment in horizontal-axis wind turbines, there have been suggestions that it might be desirable to transmit the power mechanically to the base of the tower, where electrical conversion would then take place. Although hydraulic transmission involving a controllable pump-motor pair is a possibility, considerations of pump mass and efficiency at low rpm, as well as long term reliability, indicated that mechanical transmissions are to be preferred. Of these, only the concepts involving a long vertical shaft will be considered here (chain belt drives might be lighter, but would require more servicing).

A decision on the most desirable location for the conversion equipment can only be based on a design and cost study which accounts for the accompanying modifications of nacelle and tower design. We are presently conducting such a study, and this paper will focus on the drive train dynamic implications of the choice of configuration.

Cases Studied

Figure 1 gives in schematic form the three configurations selected as representative of the range of possible arrangements. Case 1 is the standard (all aloft) configuration. Case 2 involves transmission of low speed, high torque power through a 90° bevel gear all the way down to the speed increaser and generator on the ground. In Case 3 the gearbox is aloft, and only high speed, low torque power is transmitted to the generator on the ground. Intermediate cases, with the equipment mounted halfway down the tower were also considered for cases 2 and 3.

Model Description

For purposes of dynamic analysis, a three-mass system was assumed. The inertias are those of the rotor blades plus hub, the gearbox and the generator rotor respectively (shaft inertias are neglected). The stiffnesses are those of the connecting shafts, modified in the case of the blades and the generator as described below.

The physical parameters (sizes, masses, stiffnesses) corresponding to a given configuration, rated power and mean wind speed were determined using a combination of ab-initio calculations and judicious extrapolations from the more detailed designs reported in Refs. 1, 2, and 3. The low speed shaft was assumed hollow, with internal-to-external diameter ratio of 0.54, and it was dimensioned for both torsion and bending. The high speed shafts were taken to be solid. In all cases, allowance was made for extended fatigue life.

The generator was assumed to be a 4-pole synchronous alternator, controlled for constant power angle, and hence its equivalent torsional stiffness was varied in proportion to $V^3 c_p$ $(V/\Omega R)$, where c_p is the power coefficient. No other non-linear elements were assumed.

The rotor blades were represented by its polar moment of inertia relative to the shaft, (taken to scale like $R^5 \cdot V^2_{WIND}$) and by an equivalent stiffness due to their "S-mode" lag bending. This stiffness was obtained from estimated in-plane natural frequencies, together with the blade inertias. The two blade stiffnesses were added together, and the inverse of this sum was added to the input shaft stiffness.

For calculational purposes all angular deflections, stiffnesses and intertias were reduced to the high speed side of the speed changer.

Discussion of Results

Most of the work centered on the sizes of 500 and 1500 KW, very similar in characteristics to those described in Ref. 2. Figures 2 and 3 show the variation of the drive train natural frequencies with generated power. Five cases are included for each size, namely, conventional nacelle (Case 1), long slow shaft (C2, sh = 1), intermediate slow shaft (C2, sh = 0.5), long fast shaft (C3, sh = 1), and intermediate fast shaft (C3, sh = 0.5). Only the two lowest modes are included, since the third mode has very high natural frequencies. The mode shapes for some of the cases are depicted in Fig. 4, where the deflections are normalized by the one with maximum value, and are reduced to the high speed side. Therefore, the actual deflection of the rotor disc can be obtained by dividing the amount shown by the gear ratio (of the order of 60). Mode I, with the lowest frequency, is a collective oscillation of all inertias (but mostly the rotor) resisted by the generator field and the low speed shaft. In Mode II the rotor is nearly fixed, and the other inertias oscillate against generator and slow shaft again. As discussed for instance in Ref. 4, rotational excitations arising from stationary disturbance sources can only have frequencies at even multiples of the rotation frequency. From Figs. 2 and 3 we can see that, from this point of view the standard designs (C1) are safe, since neither 2/REV nor 4/REV or even 6/REV disturbances will excite modes I or II at any power. The lowest frequency (Mode I) is below 2/REV, while the next frequency (Mode II) is at about 6/REV. The longer shafts of cases C2 and C3 have the effect of depressing both natural frequencies, with the result that mode II now becomes resonant at some intermediate power level (at 4/REV for case C3, and both at 4/REV and 2/REV for case C2). Notice that even for the full length high torque shaft, a very large stiffness reduction would still be needed to bring all of Mode II below 2/REV. This can be artificially provided by a flexible coupling, but the point is that shaft lengthening by itself is not an effective way to achieve dynamic decoupling.

This point is further emphasized in Figs. 5 and 6, which show (for the 500 KW case) torque transmissivities from the rotor disc to the generator input shaft, namely, the ratio of torque amplitudes for harmonic excitations at 2/REV (Fig. 5) and 4/REV (Fig. 6). At relatively high powers, near rated, the long shafts do achieve large reductions in the torque transmissivity, especially at 2/REV. However, this is achieved at the cost of resonances in the range below 200 KW of power. The sharpness of these resonances suggests however that with modest amounts of damping, combined with deliberate avoidance of prolonged synchronous operation at those low power levels, these resonances may not pose a fundamental obstacle.

Conclusions

(1) Long transmission shafts to ground can lead to 2 or 4/REV drive train resonances at low power levels.

(2) They do reduce greatly the harmonic torque inputs to the generator at near rated power.

References

1. "Drive Train Normal Mode Analysis for the ERDA/NASA 100 KW Wind Turbine Generator," T.L. Sullivan, D.R. Miller and D.A. Spera (NASA/Lewis) NASA TM-73718, July 1977.

2. "Design Study of Wind Turbines 50KW to 3000 KW for Electric Utility Applications, Analysis and Design," G.E., Report #NASA CR134935 February 1976.

3. "Design Study of Wind Turbines 50 KW to 3000 KW for Electric Utility Applications, Analysis and Design," Kaman Aerospace Corp., NASA Report #CR-134937, February 1976.

4. "Rotor/Generator Isolation for Wid Turbines," by L.P. Mirandy presented at AIAA-ASME 18th Structures, Structural Dynamics and Materials Conf., San Diego, CA, March 21-23, 1977.

DISCUSSION

Q. Would you comment on the accuracy of your results if you include long shaft bending modes and possible feeding of energy for torsional DOF's to bending ones (as in classical Den Hartog or Ker Wilson)?

A. The accuracy of the torsional vibration calculations is not affected by bending, provided whirling resonances are avoided. In principle, this can be done by providing intermediate bearings so that the critical shaft frequencies are raised above the shaft turning frequency. For Case 2 (long low speed shaft), a single intermediate bearing accomplishes this (especially if the shaft is made hollow). For Case 3 (long high speed shaft) operation below the first critical speed would require one bearing every 1.6 m of shaft roughly; but since lateral loads are not involved, it should be feasible to operate, say, between the second and third criticals, which implies spacings of about 4 m between bearings (or about 10 of them).

Q. Physically how would you propose a damper for the long shaft?

A. What is of interest is to damp out the generator shaft oscillations. Notice that the possible resonant conditions at 2P, 4P, etc. all affect the second mode of the drive, which involves largely generator osciallations, with little participation of the massive rotor blades. Therefore any method of introducing damping at or near the alternator should prove effective. As an example, one can estimate that for our Case 2 (long slow shaft), generator damping to the tune of 0.2 of critical (based on the isolated alternator) is sufficient to reduce the peak response at the 2P resonance to less than ±3 electrical degrees for rotor torque inputs equal to rated torque.

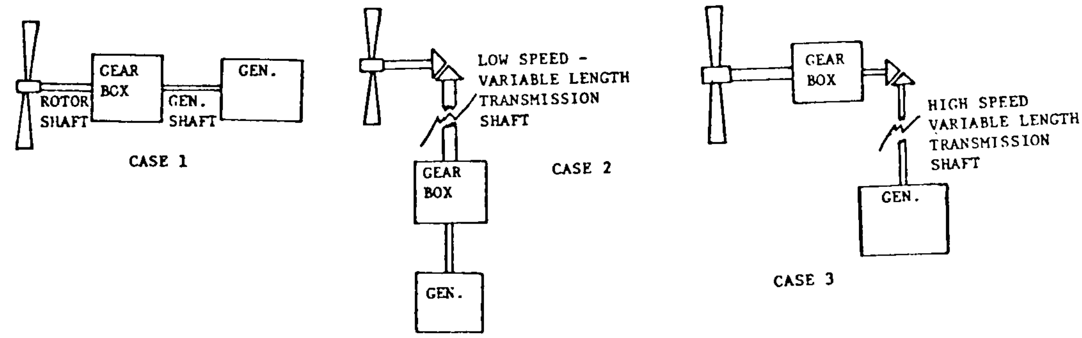

Figure 1.

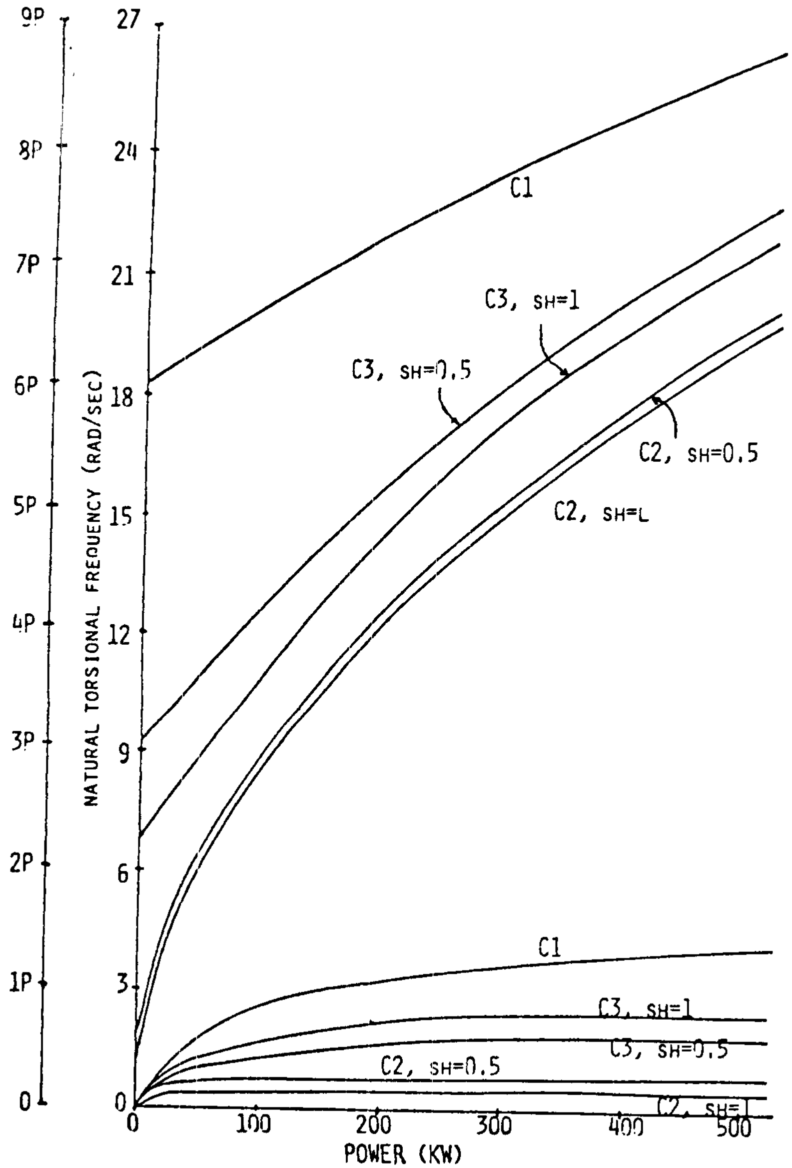

Figure 2.

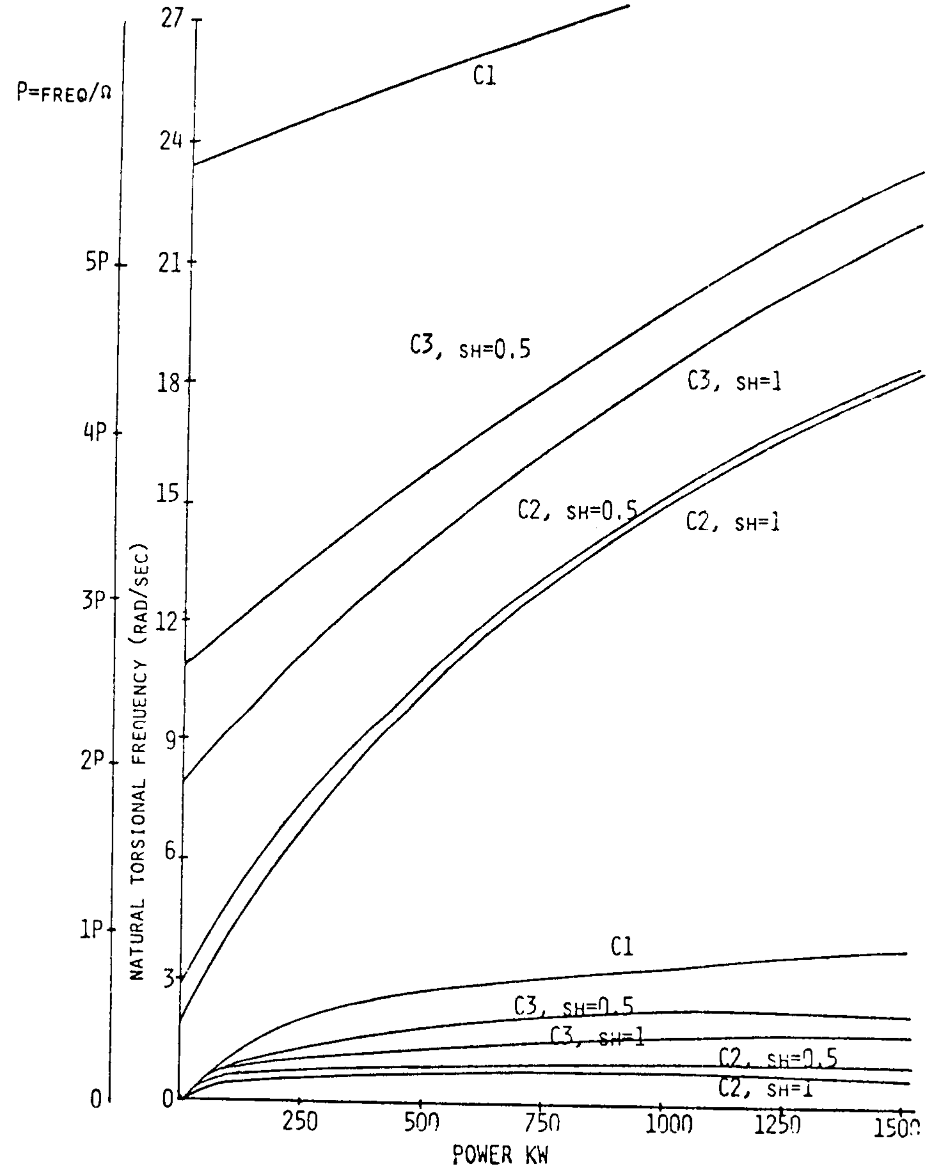

Figure 3.

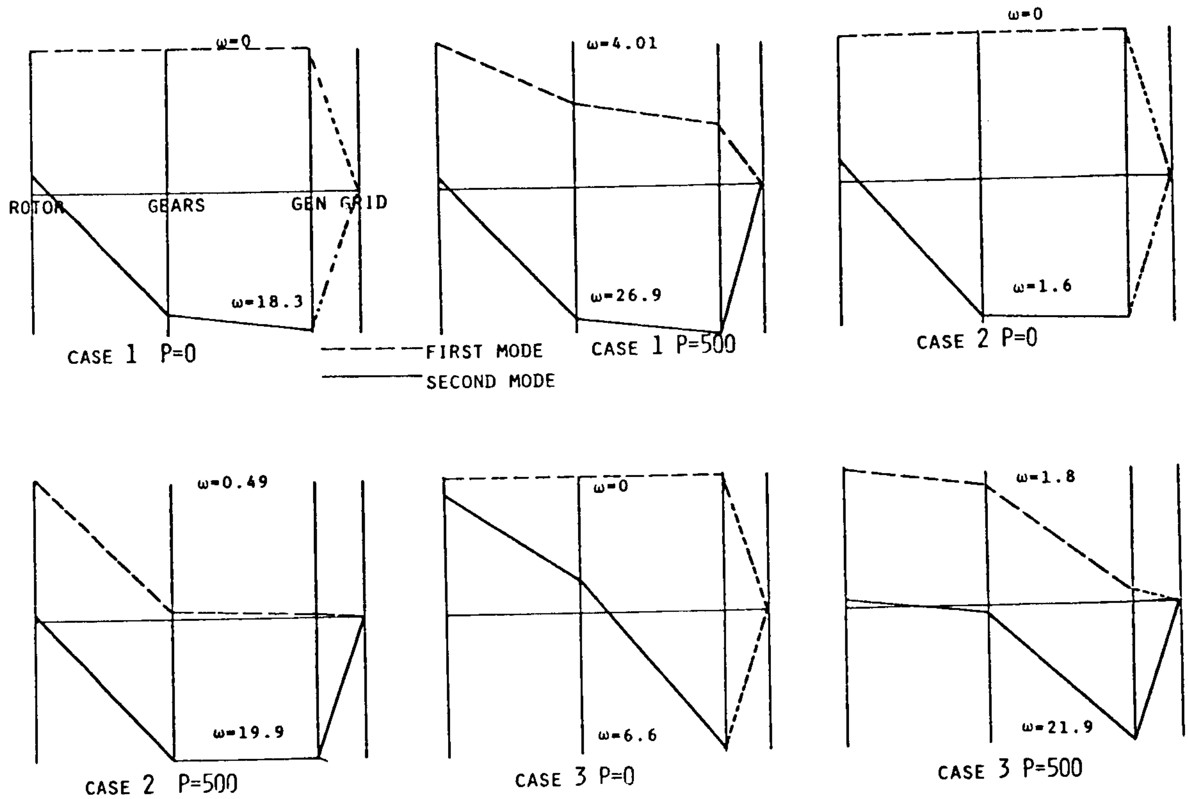
ROTOR
GEARS
GEN GRID
ω=0
ω=18.3
CASE 1 P=0
FIRST MODE
SECOND MODE
ω=4.01
ω=26.9
CASE 1 P=500
ω=0
ω=1.6
CASE 2 P=0
ω=0.49
ω=19.9
CASE 2 P=500
ω=0
ω=6.6
CASE 3 P=0
ω=1.8
ω=21.9
CASE 3 P=500

Figure 4.

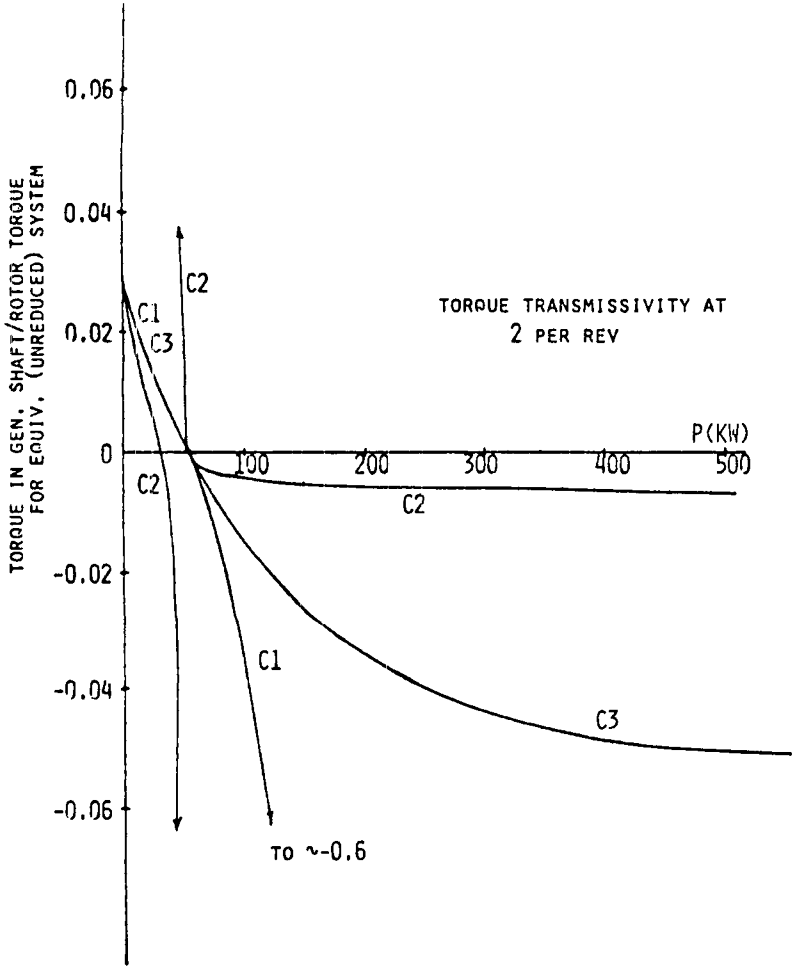

Figure 5.

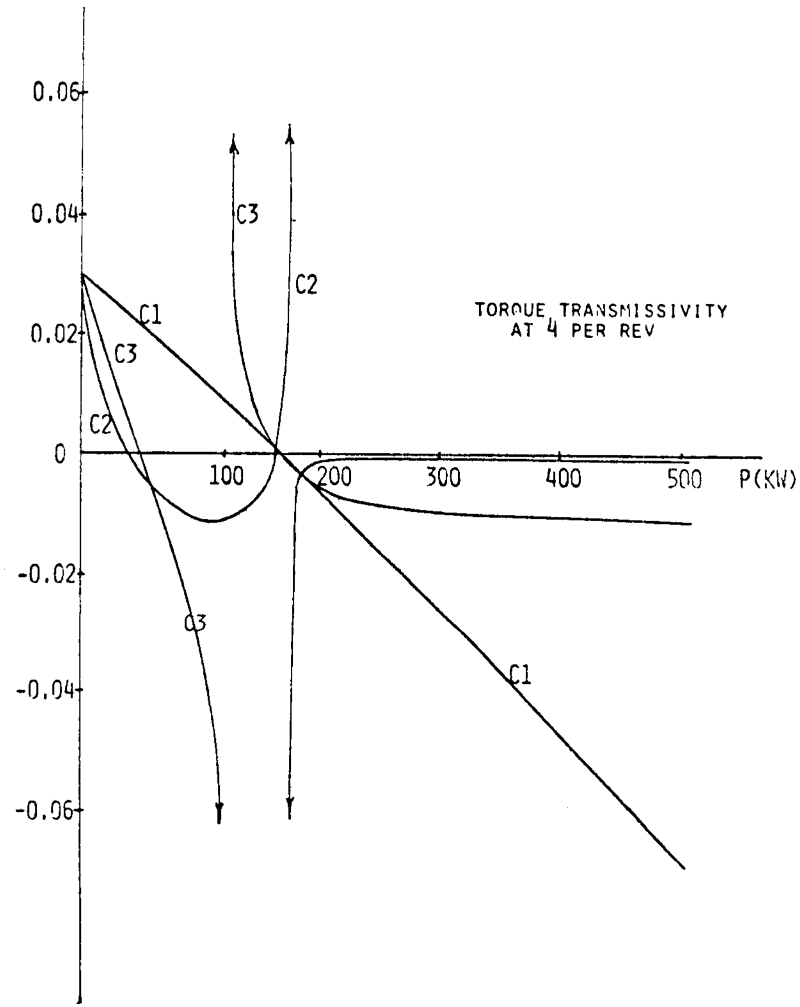

Figure 6.